9. Models for Steering Control of Motor Vehicles

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Driver control of an automobile is analyzed using feedback models for the system that consists of the driver, the vehicle, and the roadway environment. Three alternative models are evolved that satisfy the guidance and control requirements of the driver/vehicle system. They are: time-advanced lateral position, path angle plus lateral position, and heading angle plus lateral position. Although there may be other alternatives, these provide good performance, insensitivity to variations in the driver's dynamic adaptation, good predicted subjective opinion rating, etc.; and they are compatible with perceptual data from driving experiments. The resultant models give the highway and vehicle designer a tool that can be useful in analyzing the system and in optimizing its parameters for the safest and most efficient operation.

INTRODUCTION

Some form of steering control is required from the driver to make a vehicle perform a maneuver or follow the desired path. To understand driver control quantitatively it is necessary to consider the actions of random and deterministic inputs on a dynamic system comprising a vehicle whose equations of motion are known and a driver whose dynamic response under various conditions can be estimated.

Two things are needed to accomplish the driving task:

(1) A desired path (command input) that has adequate tolerances to offset variations in decision and judgmental processes

(2) A guidance and control system structure to execute the command inputs with reasonable ease and precision

Recent research (refs. 1, 2, and 3) has centered on the guidance and control area; this has included the derivation, validation, and exercise of appropriate operational models for the driver/vehicle/roadway system.

This paper is devoted to studies of steering or directional control. These are important for many reasons, not the least being that many hazardous driving situations involve steering difficulties and that this type of control is poorly understood for any phase of driving. Throttle and braking (longitudinal) control by the driver is also important, but it has been studied more extensively (e.g., refs. 4 to 10) and it is relatively well understood.

The paper begins with a short overview of the system structure to place the component elements in an overall system context. Then the components, that is, the dynamics of the vehicle and of the driver, are summarized. With the elements defined in some detail, the structure of the driver/vehicle closed-loop system is then reexamined thoroughly, considering both single-loop and multiloop forms. Finally, some implications of the theory are presented.

OVERVIEW OF THE MODEL STRUCTURE

The driver, the vehicle, and the roadway environment are the three essential parts of the closed-loop structure for a single vehicle element. The general topology of this structure for steering control is presented in figure 1. It illustrates the possible types of driver response blocks, how the driver interacts with the vehicle, and how the driver/vehicle system interacts with the roadway environment.

Three general levels of control structure are shown for the driver in figure 1. They correspond to three levels of driver behavior, and consequent system performance, and are elements of the Successive Organization of Perception in reference 11. The levels are:

(1) Precognitive which involves executing a learned maneuver in an open-loop way. It is typified by the Internally Generated Maneuver Command block. The command comes from within the driver after being triggered by some pattern or stimulus in the visual and/or proprioceptive field. Examples might include turning into one's driveway, or portions of an overtaking and passing maneuver such as the initial pullout.

(2) Pursuit which takes advantage of a knowledge of the system input to structure a driver feedforward which improves performance (ref. 12). It is shown in figure 1 as the

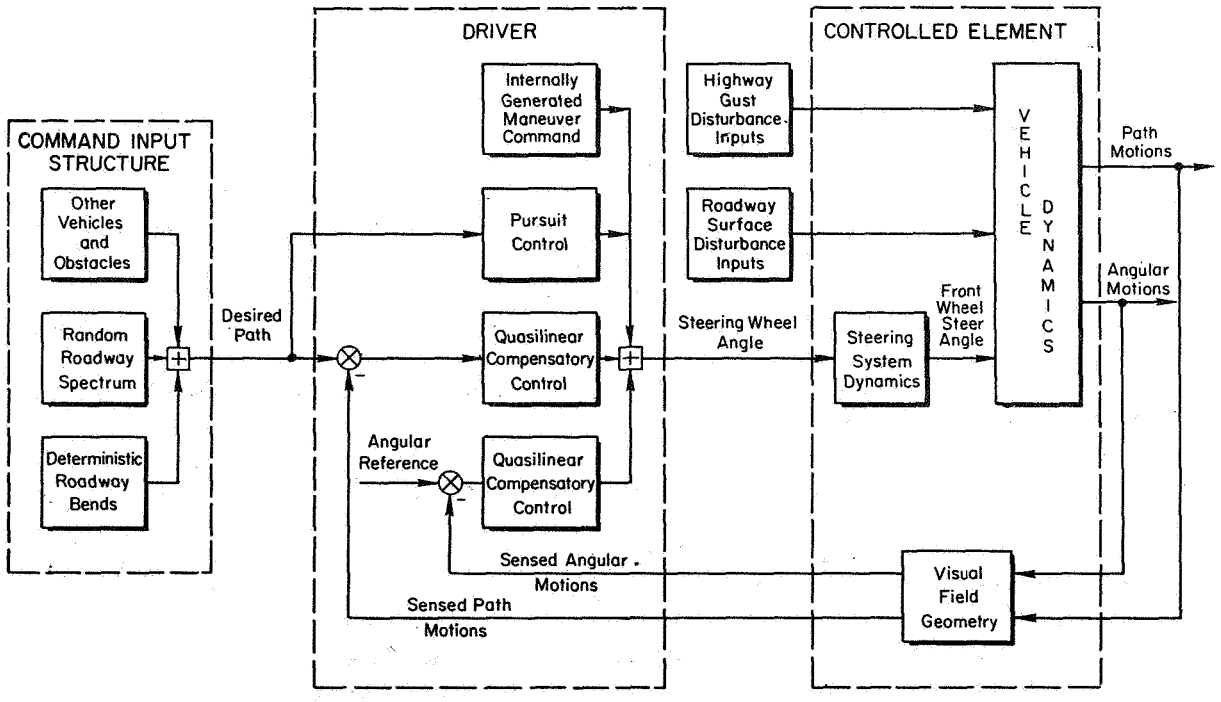

Figure 1.—Topology of the closed-loop control structure.

Pursuit Control block. The essence of pursuit behavior is the combined open-loop/closed-loop characteristic. The open-loop feedforward element provides a driver output that causes the vehicle output to very nearly duplicate the command input, while the closed-loop portion of the system acts as a vernier control to reduce any residual errors.

(3) Compensatory which implies an operation on a perceived error between the actual vehicle motion and the desired motion or input quantity. This type of control is used in the blocks labeled Quasi-linear Compensatory Control in figure 1. Compensatory differs from pursuit in that the errors only are the basis for control and command inputs are not used to structure a feedforward to the driver's output.

The levels of control activity shown are very general concepts which, nonetheless, can be applied to make predictions of system behavior. Their applicability to automobile control is limited mostly by the analyst's ability to isolate the pertinent motion error cues from the visual field geometry so that an effective controlled element can be defined.

The controlled element contains the dynamics of the vehicle and the steering system, as well as the geometry of the visual field from which the driver must extract the guidance and control cues. Driver steering control, as well as various disturbance inputs from the environment (e.g., highway gusts and roadway roughness), act on the controlled element. The resultant controlled element motions are fed back to the driver.

The roadway environment provides inputs to the system, including both commands to be followed and disturbances to be regulated against. The command input structure includes vehicles and other obstacles on the roadway, a roadway spectrum for those which curve in a random-appearing way, and discrete or deterministic roadway bends. Portions of each of these combine to give the desired path or trajectory in a given driving situation.

All the blocks in figure 1 will ultimately contain a frequency or time function. In fact, the essence of this approach is the derivation and exercise of these functions, their interaction during various driving situations, and the drawing of implications therefrom. The operational blocks (those with inputs and outputs) contain frequency functions derived from a linear or quasi-linear set of differential equations. The functions in the input blocks result from such things as spectral analyses of environmental data or time domain approximations with known deterministic forms.

Before a more specific analysis and definition of the various possible feedback loops can be made, it is necessary to have a complete understanding of the dynamic characteristics of the vehicle and the driver alone. This is accomplished below, and then attention is again focused on the closed-loop structure.

VEHICLE DYNAMICS

The dynamics of the vehicle are an important part of the effective controlled element dynamics as shown in figure 1. The driver/vehicle/roadway model examined here considers the directional or steering dynamics of the passing vehicle in three degrees of freedom. These degrees of freedom are lateral (or side) velocity v, heading angle ψ of the total vehicle, and the sprung mass (body) roll angle φ. Effects on the directional characteristics due to acceleration or braking in the longitudinal axes are included implicitly; i.e., the stability derivatives in the directional equations are computed for some known applied torque at the front and rear wheels. Road crown, grade, and superelevation are handled in a similar way by computing the vehicle equations for a nominal set of conditions or an operating point. This approach allows the use of linearized equations and analyses that allow greater insight into the control problems, yet provide good accuracy for vehicle side accelerations less than 0.3g (ref. 13).

The Laplace transformed directional equations in three degrees of freedom are given by the following matrix equation (1):

$$\begin{bmatrix} s-(Y_v+Y_{v_g}) & \frac{m_s e}{m}s^2-Y_\varphi & U_o-Y_r \\ \frac{m_s e}{I_\varphi}s-L_{v_g} & s^2-L_p s-L_\varphi & \frac{I_{\varphi z}}{I_\varphi}s-L_r \\ -(N_v+N_{v_g}) & \frac{I_{\varphi z}}{I_{zz}}s^2-N_p s-N_\varphi & s-N_r \end{bmatrix} \begin{bmatrix} v \\ \varphi \\ r \end{bmatrix} = \begin{bmatrix} Y_{\delta_w} \\ 0 \\ N_{\delta_w} \end{bmatrix} \delta_w + \begin{bmatrix} -Y_{v_g} \\ -L_{v_g} \\ -N_{v_g} \end{bmatrix} v_g \qquad (1)$$

where

- v is the lateral velocity of the unsprung mass measured at the total vehicle c.g.
- φ is the roll angle of the sprung mass about the tilted roll axis
- r is the yaw rate of the unsprung mass
- s is the Laplace transform variable
- Y_v, Y_r, Y_{δ_w} are a result of tire side force characteristics
- Y_{v_g} is a result of aerodynamic force characteristics
- Y_φ is a combination of roll steer and camber effects
- L_{v_g} is a result of aerodynamic force characteristics
- L_φ is a combination of gravity and suspension spring effects
- L_p is due to the shock absorbers and suspension friction
- L_r is equal to $-m_s e U_o / I_\varphi$
- N_v, N_r, N_{δ_w} are a result of tire side force characteristics
- N_{v_g} is a result of aerodynamic force characteristics
- N_φ is a combination of roll steer (landing to tire side force), camber side force, and suspension spring effects
- N_p is due to shock absorbers and suspension friction
- m is the total mass
- m_s is the sprung mass
- I_φ is the roll moment of inertia
- I_{zz} is the yaw moment of inertia
- $I_{\varphi z}$ is the product of inertia coupling roll and yaw
- e is the perpendicular distance from the roll axis to the sprung mass c.g.
- U_o is the forward velocity

δ_w is the mean front wheel steer angle about the kingpin

v_g is a lateral velocity gust input

The appropriate motion quantities are shown in figure 2. This matrix equation of motion provides vehicle motion quantities as perturbations of body-fixed axes. To obtain the vehicle motions in inertial space (e.g., along a roadway), it is necessary to transform the body axis motion quantities into appropriate inertial reference axes. The lateral velocity relative to an inertial coordinate system initially coincident with the unperturbed body axes is given by the integral of the lateral acceleration of the c.g., i.e.:

$$\begin{aligned} v_I(t) &= \int a_y(t)\, dt \\ &= \int [\dot{v}(t) + U_o r(t)]\, dt \\ &= v(t) + U_o \psi(t) \end{aligned} \tag{2}$$

In Laplace transform notation this becomes:

$$v_I(s) = v(s) + U_o \frac{r(s)}{s}$$

Dividing the inertial lateral velocity by U_o gives path angle γ, which is the sum of the heading angle and sideslip, i.e.:

$$\gamma = \frac{v_I}{U_o} = \beta + \psi \tag{3}$$

The lateral position relative to inertial coordinates is the integral of the inertial lateral velocity, i.e.:

$$y_I(s) = \frac{sv(s) + U_o r(s)}{s^2} \tag{4}$$

This gives the lateral position relationship between the car and the road.

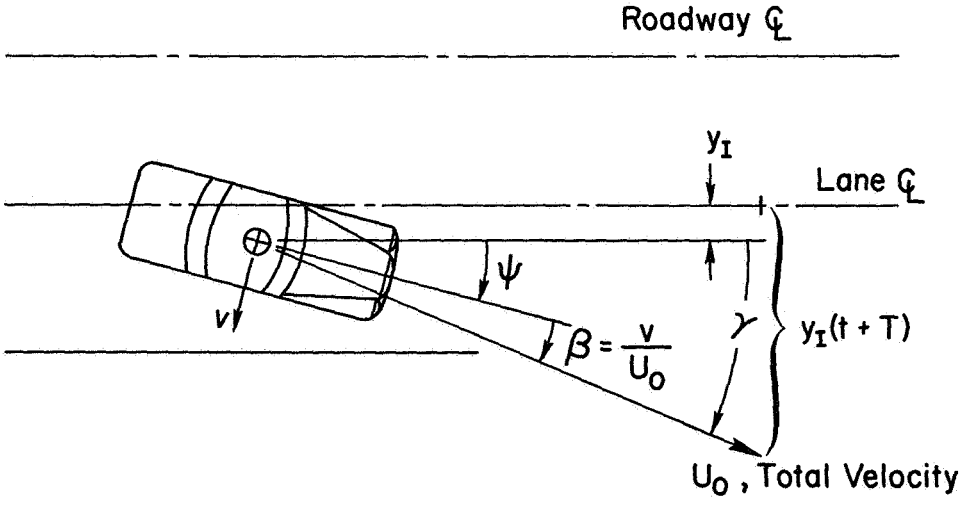

Figure 2.—Motion quantities for directional control.

Lateral acceleration at the driver's position in the vehicle, a_y, is the sum of the inertial acceleration at the axis system origin $\dot{v}_I$ plus the effect of yawing and rolling accelerations acting through moment arms to the driver's position, i.e.:

$$a_y = \dot{v}_I + l_x\dot{r} + (l_x \sin \lambda - l_z \cos \lambda)\dot{p} \tag{5}$$

where l_x is the distance in the positive x direction from the total vehicle c.g. to the driver

l_z is the distance in the positive z direction from the axis system origin to the driver

λ is the sprung mass roll axis tilt angle

Laplace transforming equation (5) and introducing $v_I(s)$ gives:

$$a_y = sv(s) + (l_x \sin \lambda - l_z \cos \lambda)\, sp(s) + (l_x s + U_o)r(s) \tag{6}$$

It is common practice in stability and control analyses to solve the matrix equation (eq. 1) to obtain ratios of motion variables to selected input quantities. The resultant ratios of polynomials in "s" are called transfer functions. Three types of polynomials are obtained—the denominator which is always the determinant of the left-hand side of equation (1), numerators which are the determinants obtained by substituting one of the right-hand column vectors into the determinant of the left-hand side, and coupling numerators which are the determinants obtained by substituting the two right-hand column vectors into the determinant of the left-hand side in various ways. The coupling numerators are used in multiloop analyses (e.g., gust regulation) to obtain the effect on outer-loop numerators of inner-loop closures utilizing another control means. Their use is discussed in detail in reference 14 and is illustrated in reference 3.

A complete summary of the vehicle dynamics pertinent to driver control is given in reference 1. It includes the derivation of the equations of motion, definition of the vehicle transfer functions, compilation of dynamic data from a number of sources, and the calculation of numerical results for a typical American sedan. Both longitudinal and directional dynamics are included, with the emphasis on the latter. Most of the details of the vehicle dynamics are omitted here, and the needed information is simply extracted from reference 1 when required.

Vehicle transfer functions have been computed in reference 1 for a typical medium-sized American sedan weighing about 4000 pounds at speeds of 30, 60, and 75 mph. The transfer functions are represented symbolically as follows:

Lateral velocity:
$$\frac{v}{\delta_w}(s) = \frac{N^v_{\delta_w}(s)}{\Delta(s)} \tag{7}$$

Heading angle:
$$\frac{\psi}{\delta_w}(s) = \frac{1}{s}\frac{r}{\delta_w}(s) = \frac{N^r_{\delta_w}(s)}{s\Delta(s)} \tag{8}$$

Roll angle:
$$\frac{\varphi}{\delta_w}(s) = \frac{N^{\varphi}_{\delta_w}(s)}{\Delta(s)} \tag{9}$$

where N(s) is a numerator polynominal in $\underline{s}$ and $\Delta(s)$ is a denominator polynominal in $\underline{s}$. The numerator and denominator polynomials for the three forward speeds are given in table 1 taken from table XI of reference 1. A fairly significant difference can be seen in the dynamics between 30 and 60 mph, while there is little difference between the results at 60

and 75 mph. This indicates that one set of directional dynamics (i.e., one operating point) can be used to analyze typical high speed driving maneuvers that involve speed changes of no more than about 15 to 20 mph.

Analysis has shown that the aerodynamic forces and moments have only a negligible effect on the steer angle transfer functions of table 1. (They are of course dominant in the gust response transfer functions, through which a highway gust disturbance input acts on the vehicle—see reference 3 where the equations and transfer functions required to introduce crosswind gusts into the driver/vehicle system are derived and summarized.)

The steering system dynamics are an important part of the vehicle dynamics and the effective controlled element. They relate the driver's steering wheel movement to the steer angle of the front wheel about the kingpin. Although the theory is well understood (e.g., ref. 15), dynamic data on contemporary steering mechanisms is very sparse. Consequently, the steering system dynamics are assumed to be a pure gain for purposes of this discussion. The effects of possible steering lags on closed-loop control are considered in reference 3.

DRIVER DYNAMIC RESPONSE

All phases of driving require some form of driver control operation. As the tasks become more demanding, the driver may change his dynamic characteristics or may alter the system structure (close other loops) to obtain the required increase in control fidelity. The driver's closure of feedback loops modifies the effective dynamics of the vehicle or controlled element and in turn determines maneuver times, stability margins, and transient response characteristics. The possible feedback loops he can introduce are determined by the sensory information available.

An introductory discussion of driver response characteristics was illustrated in figure 1, and several possible types of driver response were presented: quasi-linear compensatory control, pursuit control, and internally generated maneuver commands. The first requires a fairly complex description, but it is well understood at the current time and probably comprises a significant portion of the driver's active control efforts. The remaining two are somewhat easier to describe qualitatively, but predictive models for these processes are not yet well developed.

The quasi-linear blocks in the driver/vehicle model are most appropriate for defining his response to random-appearing external inputs. This includes command inputs due to the bends and curves in the roadway or desired path, as well as disturbance inputs due to gusts and roadway roughness. These quasi-linear blocks are relatively quiescent in the presence of deterministic inputs when the other types of response dominate. They are active during pursuit control if there are also random-appearing disturbances present, which the driver cannot preview.

QUASI-LINEAR COMPENSATORY CONTROL CHARACTERISTICS.—The quasi-linear describing function model of the operator has resulted from an exhaustive series of human operator dynamic response measurements made over a period of about two decades, culminating at present in references 16 to 19, for example.

This model consists of a describing function component with parameters that depend on the system and situation, an additive remnant, and a set of adjustment rules that tell how to adjust the describing function parameters. This quasi-linear model is depicted in the illustrative single-loop block diagram of figure 3.

In its most complete form the describing function contains a gain, an indifference threshold, a time delay, an equalization characteristic, and high frequency neuromuscular

TABLE 1.—TRANSFER FUNCTIONS FOR TYPICAL AMERICAN SEDAN

Polynomial	30 mph	60 mph	75 mph
Lateral velocity numerator, $N^{v}_{\delta_W}(s)$	$180(s+.188)[s^2+2(.305)(6.38)s+(6.38)^2]$	$180(s-9.50)[s^2+2(.284)(6.94)s+(6.94)^2]$	$180(s-13.1)[s^2+2(2.75)(7.00)s+(7.00)^2]$
Yaw rate numerator, $N^{r}_{\delta_W}(s)$	$24.3(s+9.49)[s^2+2(.278)(6.66)s+(6.66)^2]$	$24.3(s+4.35)[s^2+2(.284)(6.95)s+(6.95)^2]$	$24.3(s+3.40)[s^2+2(.279)(7.03)s+(7.03)^2]$
Roll angle numerator, $N^{\varphi}_{\delta_W}(s)$	$-49.04[s^2+2(.570)(6.85)s+(6.85)^2]$	$-49.04[s^2+2(.284)(6.85)s+(6.85)^2]$	$-49.04[s^2+2(.228)(6.85)s+(6.85)^2]$
Lateral acceleration at origin numerator, $N^{a_{y_o}}_{\delta_W}$	$180[s^2+2(.194)(6.73)s+(6.73)^2][s^2+2(.499)(7.43)s+(7.43)^2]$	$180[s^2+2(.293)(6.91)s+(6.91)^2][s^2+2(.160)(7.24)s+(7.24)^2]$	$180[s^2+2(.279)(6.77)s+(6.77)^2][s^2+2(.125)(7.39)s+(7.39)^2]$
Characteristic denominator, $\Delta(s)$	$.684[s^2+2(.296)(7.09)s+(7.09)^2][s^2+2(.986)(9.17)s+(9.17)^2]$	$.684[s^2+2(.785)(5.84)s+(5.84)^2][s^2+2(.290)(6.97)s+(6.97)^2]$	$.684[s^2+2(.670)(5.306)s+(5.3)^2][s^2+2(.309)(6.92)s+(6.92)^2]$

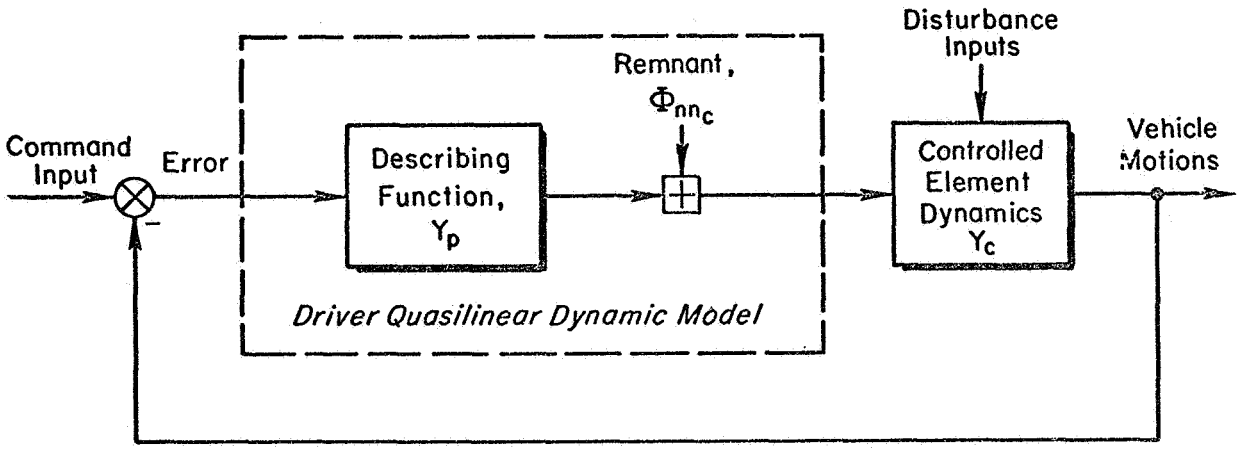

Figure 3.—Driver control in compensatory task.

system dynamics. The indifference threshold is a higher order effect that can often be ignored when the inputs are large, and under other conditions it can be accounted for by using decreased driver gain. The neuromuscular system dynamics are based on very low and very high frequency data, and can be approximated at the midfrequencies of interest in driving as a first-order lag or even as an added increment to the time delay. With these simplifications, the general driver describing function of reference 3 reduces to:

$$Y_p \doteq \frac{K_p e^{-j\omega\tau}(T_L j\omega + 1)}{(T_I j\omega + 1)(T_N j\omega + 1)} \tag{10}$$

or, alternatively,

$$Y_p \doteq K_p\left(\frac{T_L j\omega + 1}{T_I j\omega + 1}\right) e^{-j\omega(\tau + T_N)} \tag{11}$$

where K_p is the gain

$\left(\frac{T_L j\omega + 1}{T_I j\omega + 1}\right)$ is a simplified equalization characteristic

τ is the time delay

T_N is the neuromuscular system time constant

The variable $j\omega$ has been used in place of $\underline{s}$ to indicate that the describing function is most appropriate when the inputs approximate stationary random processes. The form of equation (11) is completely adequate for typical driver/vehicle closed-loop analyses involving reasonably stable vehicles. Thus, the describing function form reduces to a transfer-function-like element containing a gain, an adjustable equalization, and an exponential term

that affects only the phase angle. It is valid for a variety of drivers, inputs, vehicle dynamics, steering system characteristics, and loop structures. Most of the parameters are adjustable as needed to make the vehicle motions follow the command input and regulate against the disturbance input. Experimental measurements of all the parameters are reported in reference 3 and summarized in succeeding paragraphs.

The pure time delay represented by the $e^{-j\omega\tau}$ term is due to sensor excitation (the retina in the visual case), nerve conduction, computational lags, and other data processing activities in the central nervous system. It contains components that are closely related to certain kinds of classical reaction times. A portion of the time delay is currently taken to be a constant, because it appears to be essentially invariant with the input and vehicle dynamics for either single or dual random-appearing input tasks. However, both intersubject and intrasubject variations occur, and observed values of the constant part of τ range from about 0.05 to 0.125 second, with a nominal value of about 0.10 second (ref. 16). The remaining component of the pure time delay is an increment present when low frequency lead equalization is generated by the driver. For instance, with first-order lead equalization the base value of τ is increased by about 0.15 second.

The neuromuscular time constant T_N is partially adjustable for the task. The nature of the adjustment is input-adaptive, consisting of a monotonic decrease in T_N with increasing forcing function bandwidth ω_i. The data of reference 16 show that the incremental reduction is approximately

$$\Delta T_N \doteq 0.08\omega_i \quad , \quad \omega_i \leq 4 \text{ rad/sec} \tag{12}$$

The observed variation of T_N with forcing function bandwidth ranges from less than 0.1 second to almost 0.5 second. In cases where the input bandwidth is not known, typical values of T_N near 0.1 second are sometimes used.

The equalizing characteristics, $(T_L j\omega + 1)/(T_I j\omega + 1)$, coupled with gain K_p are the major adaptive elements of the human that allow him to control many differing dynamic devices. Their function is the modification of the stimulus signal into a suitable neuromuscular command which is properly scaled and phased for proper overall man/machine system operation. For given input and vehicle dynamic characteristics, the form of the equalizer is adapted to compensate for the vehicle dynamics and the driver's pure time delays. The major cost of equalization is the increase in time delay incurred when low frequency lead is needed as part of the compensation.

The major adjustment rules for the equalization characteristics, evolved as generalizations of many experiments, are that a particular equalization is selected from the general form $K_p(T_L j\omega + 1)/(T_I j\omega + 1)$ so that the following properties are attained:

(1) The driver/vehicle system can be stabilized by proper selection of gain, preferably over a very broad region.

(2) The amplitude ratio of the product of the driver describing function and the vehicle dynamics $|Y_p Y_c|$ has approximately a -20-decibel/decade slope in the crossover region—that frequency band centered on the crossover frequency ω_c.

(3) $|Y_p Y_c| >> 1$ at low frequencies to provide good low frequency closed-loop response to system commands and good suppression of the effects of disturbances.

The region near the crossover frequency ω_c (where $|Y_p Y_c| \doteq 1$) has a surpassing importance in the synthesis of feedback systems. First, for good performance the crossover frequency must exceed the largest frequency ω_i at which there are appreciable components of the commands or external disturbances. In this sense, the crossover frequency amounts to an upper bound on the frequency region wherein $|Y_p Y_c| >> 1$, a condition that

provides good following of the system input by the output and suppression of disturbance effects. Second, the shape of Y_pY_c at and near the crossover frequency determines the dynamics of the dominant modes of the system response. The conditions for neutral oscillatory stability are

$$|Y_pY_c| = 1 \text{ and } \angle Y_pY_c = -\pi \text{ (in radians)} \tag{13}$$

when sign $Y_pY_c(0)$ is plus. Thus, neutrally stable or unstable dominant modes are usually avoided if the system satisfies the condition

and

$$|Y_pY_c| < 1 \quad \text{when} \quad Y_pY_c = -\pi$$
$$\angle Y_pY_c > -\pi \text{ when } |Y_pY_c| = 1$$

These are the often quoted criteria of positive gain margin and phase margin. For simple minimum phase or minimum phase plus transport lag systems, they are an expression of the Nyquist stability criterion. Because stability is a fundamental requirement in any feedback control system, these requirements for gain and phase margin may severely restrict the choice of the crossover frequency.

The human corresponds fairly well to the criteria for a good controller, varying somewhat with vehicle dynamics and input bandwidth. Experimental values of crossover frequencies ω_c and phase margins φ_M for several dynamic forms and bandwidths are given in table 2, abstracted from reference 16. These data indicate the invariance properties of ω_c, which are:

ω_c—K_c independence.—After initial adjustment, changes in vehicle gain K_c are offset by changes in driver gain K_p so that system crossover frequency ω_c is invariant with vehicle gain.

TABLE 2.—PHASE MARGINS AND CROSSOVER FREQUENCIES

Y_c	ω_i = 1.5 rad/sec		ω_i = 2.5 rad/sec		ω_i = 4.0 rad/sec	
	φ_M (rad)	ω_c (rad/sec)	φ_M (rad)	ω_c (rad/sec)	φ_M (rad)	ω_c (rad/sec)
K_c	0.51	5.1	0.75	5.7	0.75	6.7
K_c/s	0.42	4.6	0.73	4.7	0.94	5.0
$K_c/(s-2)$	0.35	4.6	0.66	5.0	0.70	5.2
K_c/s^2	0.26	3.2	0.51	3.3	0.73	1.8

ω_c—ω_i independence.—System crossover frequency depends only slightly on input bandwidth for $\omega_i < 0.8\ \omega_{c_o}$, where ω_{c_o} is the ω_c for a given Y_c extrapolated to zero input bandwidth ($\omega_i = 0$).

ω_c regression.—When ω_i nears or becomes greater than $0.8\ \omega_{c_o}$, the crossover frequency regresses to values much lower than ω_{c_o}.

Phase margin adjustment with ω_i.—Because ω_c is essentially independent of ω_i and ΔT_N is directly proportional to ω_i, the system phase margin φ_M is directly proportional to ω_i. This strong dependence of phase margin on the input bandwidth is associated with the linear variation of ΔT_N with ω_i and is essentially an alternate statement of equation (12).

The crossover frequencies presented in table 2 are maximum values for skilled subjects in fixed-base simulators without motion feedbacks. These will ordinarily be considerably reduced during driving, especially when full attention is not demanded. The maximum extent of this reduction can be readily estimated for conditionally stable systems on the basis of stability alone. To obtain an estimate of the full-attention driving value for vehicles that are always stable, consideration should be given to:

(1) Gain reduction

(2) τ_e increase, due to conflicting demands

(3) τ_e increase, due to steering dynamics, especially when holding trim loads against road crown or steady crosswinds

(4) Statistical variation in ω_c with time and between subjects

This leads, in practice, to larger phase and gain margin criteria, typical examples of which are used in the driver/vehicle loop closures shown below and in reference 3.

In many driving situations the driver's regulation or control activity is only intermittent, so the average crossover frequencies will invariably be less than those estimated for full-attention driving. This reduction is probably due primarily to an increase in indifference threshold, and thus may not result in much of a reduction for large amplitude motions. A matter of some importance is the spread between the highest and lowest ω_c values possible (the lowest often merges with the unattended condition), which is a measure of the degree of vehicle configuration forgiveness. This or an associated measure also implies limitations on indifference threshold, sampling, minimum and maximum average movements per second, minimum and maximum information rates, etc.

Simplified Crossover Model for the Driver Describing Function.—Many of the preceding remarks about the rationale of equalization adopted by the driver can be simplified by using the approximate crossover model. The experimental data and consideration of the requirements of good feedback system performance both lead directly to the conclusion that the driver adjusts his describing function so that the open-loop function Y_pY_c in the vicinity of the gain crossover frequency ω_c is closely approximated by

$$Y_pY_c \doteq \frac{\omega_c e^{-j\omega\tau_e}}{j\omega} \tag{14}$$

where τ_e is an effective pure time delay that includes the neuromuscular time constant T_N as well as τ and any net high frequency controlled element lag. This crossover model is not a replacement for the more complete model, but it is a convenient engineering approximation. While it is a better description of amplitude ratio characteristics than of phase characteristics, it often describes the most significant features of operator behavior adequately. This is because the actual shape of the open-loop function away from the gain crossover frequency is usually almost irrelevant to the closed-loop performance. It is, therefore, often unnecessary to retain terms in the describing function whose influence is not felt in the immediate vicinity of gain crossover.

Table 3 shows the application of the adjustment rules to the prediction of a driver's describing function and to the formulation of the crossover model for several simple (limiting-case) controlled element dynamics. In many instances more complicated controlled elements can be satisfactorily approximated in the vicinity of gain crossover by these simple limiting cases.

Driver Remnant Characteristics.—The remnant is that portion of the driver's output which is not linearly correlated with the input. It is always present to some extent, and it is probably the most significant undesired input into the steering system from a practical standpoint. The remnant is considered to be a random process added to the output of the driver's describing function block to form the total driver's output. It is denoted by a power spectral density Φ_{nn} in figure 3. The point of application could be moved to other places in the loop as long as no nonlinear elements are passed in the process. The remnant can have significant power at frequencies that are high enough to excite lightly damped high frequency steering system modes present under conditions of low Coulomb friction (e.g., ref. 15).

The major source of remnant appears to be nonstationarity in the operator's behavior (ref. 16), manifest as time-varying components in gain K_p and effective time delay τ_e. The remnant component due to τ_e variation is usually increased when low frequency lead is generated by the driver. Also, low frequency lead is associated with a pulsing behavior by the operator in which his output tends to be pulses with areas roughly proportional to the stimulus amplitude. This is an additional remnant source for controlled element dynamics that require low frequency lead equalization. An extensive body of remnant data shows that the remnant power increases with the order of the controlled element dynamics and with increases or decreases in controlled element gain away from an optimum value. It decreases with input bandwidth.

For nominally good dynamics (e.g., $Y_c \doteq K_c/s$) the remnant power is about 30 decibels down relative to the input. Consequently, the remnant can usually be neglected from the standpoint of predicting driver/vehicle closed-loop response characteristics when the vehicle characteristics are reasonably good. This assumption is made in conventional analyses as a starting point. Then, if there is reason to believe that significant remnant power is present in the vehicle output motion quantities, the remnant can be added to the output of the driver/vehicle system.

Driver Response in Multiple-Loop Situations.—The closed-loop structure of figure 1 involving angle and path feedbacks is called multiloop single-point control. Multiloop implies two or more dynamically coupled motion variables, and single-point refers to one control means, the steering wheel. Experimental measurements of operator response in such a multiloop situation (ref. 19) showed that his describing function in the outer loop can be obtained by application of the single-loop model described above. Thus, the equalization in an outer loop involving lateral position is obtained by direct application of the describing function model once the appropriate inner loop has been closed.

Ordinarily the inner feedback loops supplied by the driver act as parallel equalization for the outer loop, or provide feedbacks or crossfeeds that suppress subsidiary controlled element degrees of freedom that have undesirable effects on subsequent loops. Because the role of the inner loops is so dependent on outer-loop requirements, the rules cited above for the single-loop model are not generally applicable, e.g., even stability of an inner loop may not be required. The types of inner loops closed and the equalization selected should be compatible with one or all of the following considerations:

(1) Outer-loop adjustments per the single-loop adjustment rules become more feasible, e.g., $|Y_pY_c|$ for the outer loop can be made approximately -20 decibels/decade with less outer-loop equalization by the driver.

TABLE 3. — DRIVER DESCRIBING FUNCTION AND CROSSOVER MODEL

WHEN THE EFFECTIVE CONTROLLED ELEMENT TRANSFER FUNCTION IS (Y_c)	THE DRIVER'S APPROXIMATE DESCRIBING FUNCTION IS (Y_p)	THE CROSSOVER MODEL IS (Y_pY_c)	EFFECTIVE TIME DELAY FOR $\omega_1 = 0$ $[\tau_e(0)]$ (sec)
$\frac{K_c}{s} = \frac{K_c}{j\omega}$	$K_p e^{-j\omega\tau_e}$	$\frac{K_p K_c e^{-j\omega\tau_e}}{j\omega}$	1/3
K_c	$\frac{K_p e^{-j\omega\tau_e}}{(T_I j\omega + 1)}$; $\frac{1}{T_I} \ll \omega_c$	$\frac{K_p K_c}{T_I} \frac{e^{-j\omega\tau_e}}{j\omega}$	1/3
$\frac{K_c}{s^2} = \frac{K_c}{(j\omega)^2}$	$K_p e^{-j\omega\tau_e}(T_L j\omega + 1)$; $\frac{1}{T_L} \ll \omega_c$	$\frac{K_p K_c T_L e^{-j\omega\tau_e}}{j\omega}$	1/2

(2) The sensitivity of the closed-loop characteristics to changes in either inner- or outer-loop driver characteristics is reduced from that in an outer-loop-only situation. This includes the improvement of stability margins.

(3) The loop structure and equalization selected are those for which total subjective opinion rating is the best obtainable.

It frequently happens that the driver describing function synthesized for the inner loop alone via the single-loop model is also the one that best enhances the outer-loop closures. This is most common in situations where the basic vehicle possesses good stable dynamic characteristics, and the control task is merely following command inputs or suppressing disturbances — not stabilizing the vehicle.

PURSUIT CONTROL CHARACTERISTICS.—The pursuit control block in figure 1 operates on the input using the driver's preview of a desired path, rather than a perceived motion or path error as in the compensatory case. Recent experiments (ref. 13) have shown that the describing function magnitude of the driver's pursuit feedforward block Y_{p_i} is approximately equal to the inverse of the magnitude of the effective controlled element dynamics, i.e., $|Y_{p_i}| = |1/Y_c|$. The net effect is to make the amplitude of the total closed-loop describing function from the command path to actual vehicle path be approximately unity (although the data show some variations in phase angle), and the commanded path and the actual path are approximately equal. The driver must be able to see and use the input itself to structure this block. The compensatory loops can be active when the pursuit loop is operative, providing, for example, vernier corrections and regulation against disturbance inputs that are most evident in the error.

INTERNALLY GENERATED MANEUVERS.—When a discrete disturbance (e.g., an isolated gust) is encountered, the change in the vehicle motion quantities is perceived by the driver as a motion error. The well-trained driver will recognize that the disturbance has a deterministic form for which there is some appropriate skilled response. If the input is a step, for example, he may make a classic response (e.g., ref. 21) involving a time delay, a relatively rapid steering wheel motion during a rise time phase, and an error-correcting phase. Recent operator response modeling activities (ref. 22) have concentrated on deriving and refining models for discrete inputs (such as steps and ramps) with various effective controlled element dynamics. Some potentially useful results have been obtained, but their application to the prediction of driver behavior is not yet state of the art. Any discrete response block is assumed to be quiescent in the presence of low frequency random motions, and to respond only to simple deterministic signals. During such a response the other driver blocks are assumed to be relatively inactive.

A number of the control actions made by a practiced driver during various phases of driving are done in a precognitive (or open-loop) manner. A good example is pulling out to pass and pulling back in. In this case the driver produces the appropriate steering action based on an internally generated pattern previously evolved during a learning process. He is cognizant of the effective controlled element, the command input, and the state of the motion quantities reflected in the error; but is not operating on any motion quantity in a linear or quasi-linear way. Safety aside, he could in effect close his eyes and complete the maneuver, leaving some residual position error at the end. The patterns produced may result in nearly optimal response according to some (yet undefined) criterion such as minimum time, minimum overshoot, etc. The essence of this block for maneuver command generation is that it produces whatever control signal is required to accomplish the desired maneuver to some degree of accuracy acceptable to the driver.

STRUCTURES OF DRIVER/VEHICLE CLOSED-LOOP SYSTEMS

Perhaps the most difficult problem in driver/vehicle closed-loop analysis is to determine what sensory feedback loops the driver is using. There are two general approaches to deriving the closed-loop structure: from consideration of perception and from guidance and control theory. The perceptual basis relies on in situ experimental observation of subjects driving, and can require extensive instrumentation, eye-movement cameras, highly structured and constrained experimental environments, etc. The primary aim is to discover experimentally what cues are used to drive. The control theory approach is to consider the driver/vehicle system as a guidance and control problem, and then to develop the kinds of vehicle motions that must be sensed and commands that must be inserted to satisfy the guidance and control needs. The guidance and control problem does not have a unique solution, so the initial results of this kind of study are a number of sufficient systems. Consideration must then be given to the driver's ability to perceive (sense) the vehicle motion and input quantities and to the nature of the operations required on the sensed quantities in closing the driver/vehicle system loops. Those potential systems that involve readily sensed quantities and relatively simple (e.g., proportional, minimum conscious effort, etc.) operations on the sensed motion variables are then accepted as good candidates. The control theory approach is relatively straightforward, given the current state of knowledge of typical operator/vehicle control laws and closed-loop analysis techniques. The best approach is to marry the guidance and control and the perceptual theories, for the net results must be of necessity compatible, and the two viewpoints offer much to one another as collaborators and corroborators. However, it has been necessary to rely on the control theory approach as the main tool here because of the limited amount of perceptual data relevant to vehicular control.

SINGLE-LOOP STRUCTURE.—The problem of how the driver perceives the feedback variable is secondary in the deductive guidance and control approach. The objective is to determine how well the variable permits the driver to control the vehicle if it can be sensed. This is pursued by attempting to discover likely feedback loops that may exhibit good characteristics as either a command loop or as a subsidiary loop that provides equalization for a command loop. It is also desired to identify those loops which possess poor characteristics as single-loop systems that either rule them out as possible feedbacks or indicate the need for inner-loop equalization or that exhibit properties that make them candidates as accident causes if they are inadvertently opened or closed.

A systematic search for likely feedback loops has been accomplished (ref. 3) using the dynamics from reference 1 of a typical sedan (circa 1965) and the driver describing function characteristics. The search took the form of a survey of single-loop closures directed toward determining those loops which the driver might close. The quality of the resultant closures was judged by

(1) The equalization required by the driver to provide a stable system consonant with the adjustment rules

(2) The closed-loop performance attainable as measured by crossover frequency, stability margins, etc.

(3) The subjective opinion the driver would have of his control task (as determined by the amount of lead equalization he must adopt, precision of describing function adjustment required, etc.)

The closures were restricted to compensatory mechanizations. This is valid and consistent with such concepts as preview when one considers that the effective controlled

element dynamics include the geometry and kinematics of the visual field as well as the dynamics of the vehicle. In essence, a guidance or control cue is perceived someplace in the visual field by the driver, and to the extent that the driver steers the vehicle to modify (e.g., follow or reduce) this cue in some way, he is acting in a compensatory manner.

The results of driver/vehicle single-loop surveys are summarized in table 4. The same closure criterion was used for each of the loops in table 4 to maximize the validity of relative comparisons. Also, the same effective time delays were used in all systems for computational convenience—this has the effect of making the crossover frequency for those systems with driver-generated low frequency lead larger than it would be with a more precise estimate. The absolute levels shown will vary from one driver to another, depending on skill, attention, fatigue, etc.; but these levels should all change in roughly the same way as a function of these factors for a given driver.

Closure of an inertial lateral deviation loop is very likely necessary to stay in the lane or to follow a desired trajectory. The driver is assumed to be steering according to his lateral deviation with respect to an inertial axis reference such as the roadway or lane centerline. Although the driver can easily perceive this motion quantity, this single-loop control mode is relatively difficult. The effective controlled element transfer function is given by:

$$\frac{y_I}{\delta_w} = \frac{180\left[s^2 + 2(0.160)(7.24)s + (7.24)^2\right]\left[s^2 + 2(0.293)(6.91)s + (6.91)^2\right]}{0.684s^2\left[s^2 + 2(0.785)(5.84)s + (5.84)^2\right]\left[s^2 + 2(0.290)(6.97)s + (6.97)^2\right]} \quad (15)$$

The system survey plot is shown in figure 4. The system is unstable with a pure gain plus time delay (unequalized) driver describing function, as shown by the solid lines, because of the two poles at the origin (see root locus). A relatively large amount of driver lead equalization is required to attain even minimal stability margins. A skilled driver could add low frequency lead at about 0.5 rad/sec and obtain a crossover frequency of about 1.0 rad/sec with 30° of phase margin. This is shown by the dashed lines in figure 4, and corresponds to a driver model of the form

$$Y_{p_{y_I}} \doteq \frac{K_{p_{y_I}}}{0.5}(s+0.5)e^{-0.3s} \quad (16)$$

Although a skilled driver could achieve the lead equalization of equation (16) for a short period of time, it would be difficult and he would have a poor subjective opinion of the vehicle's dynamics (ref. 23). Thus, an inertial lateral deviation loop has poor characteristics as a single-loop system, and plainly requires additional (inner) loop closures to alleviate the need for large driver lead to obtain a satisfactory driver/vehicle system bandwidth.

Although $y_I \rightarrow \delta_w$ is not a likely primary control loop because of the low bandwidth and high driver skill required, it may be important after perceptual transitions. This could occur when the driver suddenly loses his view down the road and is forced to steer on the basis of his lateral position in the lane (e.g., entering a tunnel or fog bank). Such a regression would cause him to slow down so that the lower driver/vehicle system bandwidth would still enable him to follow any likely command input.

Both heading angle and path angle are control feedbacks that offer good closed-loop characteristics. (Detailed surveys are shown in figs. 8 and 10, respectively.) Path angle

TABLE 4.—SUMMARY OF SINGLE-LOOP SURVEYS

FEEDBACK	DRIVER EQUALIZATION	ESTIMATED SYSTEM CROSSOVER FREQUENCY (rad/sec)	ESTIMATED RELATIVE DRIVER OPINION
Inertial lateral deviation $y_I \rightarrow \delta_w$	Large lead	1	Poor, because of large amount of lead equalization and low bandwidth
Heading angle $\psi \rightarrow \delta_w$	Gain only	1.5	Good
Path angle $\gamma \rightarrow \delta_w$	Gain only	1	Good, particularly with small lead equalization (ref. 3)
Lateral velocity $v \rightarrow \delta_w$	Lag-lead	1	Fair, because of low bandwidth and lag equalization
Heading rate $r \rightarrow \delta_w$	Lag-lead	2.5	Good
Lateral acceleration at driver's head $a_y \rightarrow \delta_w$	Lag-lead	1	Fair, because of low bandwidth, lag equalization, and high sensitivity

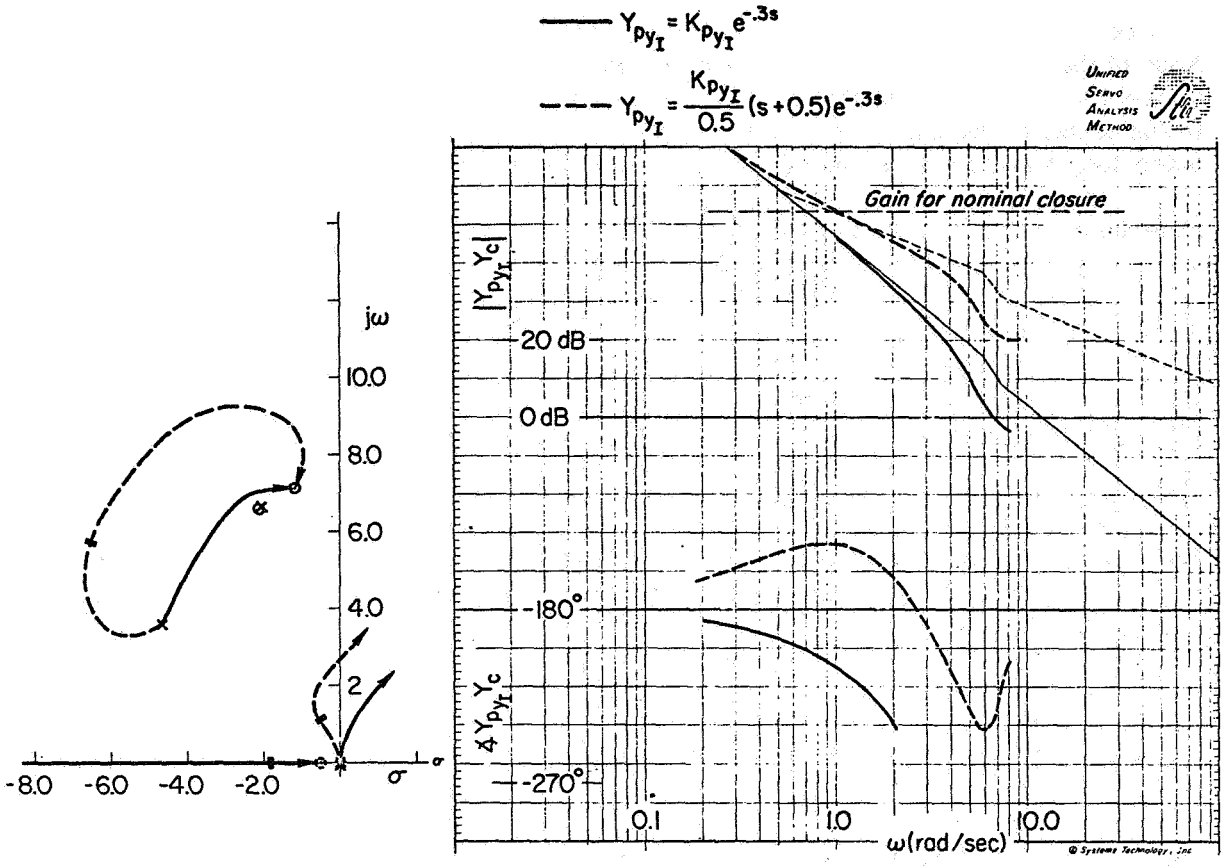

Figure 4.—Inertial lateral deviation loop closure.

control combines heading control (a good loop by itself) with body axis lateral velocity (sideslip), which is only fair taken alone. Both heading angle and path angle systems can serve as outer loops if an intermittent trim loop is employed to occasionally reduce or reset the lateral deviations. Also (as described later), either of these angle systems can serve as an inner loop to reduce the lead requirements for an inertial lateral deviation outer-loop system.

The heading rate system is also good and is a conceivably useful inner loop. Path angle rate (not shown) has similar potential. On the other hand the lateral acceleration at the driver's head (a quantity related to path angle rate) is not a particularly good system because it is highly sensitive to driver gain variations; that is, the difference between the minimum gain required to provide some control and the maximum gain permissible without instability is not large. This also places an additional requirement on the need for relatively high driver skill.

The single-loop closures in table 4 that have poor characteristics are unlikely loop closures under favorable conditions because of the poor system performance or excessive driver demands. They can, however, be important in situations where a better loop structure has been destroyed due to changes in the cues available or other disturbing factors. Under such perceptual transitions, these poor single-loop systems may momentarily exist as a transitional phase.

MULTILOOP STRUCTURES.—The good single-loop closures shown in table 4 will all provide systems with good performance in following a command input of that motion variable. None of the single-loop closure systems shown will do a very good job of following a path or trajectory command input that involves minimizing lateral position (deviation) errors to stay in the center of the lane or roadway. Thus, the need exists for the driver to augment his outer-loop structure with inner loops that serve as appropriate equalization. Among these multiloop systems, those which require little or no driver equalization (i.e., only gain plus time delay in each of the loops) are to be preferred from both a closed-loop performance and a driver subjective opinion standpoint.

Three different multiloop systems have evolved to date. Their block diagrams are shown in figure 5, and their characteristics are summarized in table 5, based on the detailed system survey analyses of the appendix. Each of the selected multiloop configurations is discussed briefly below in their nominal good configuration. Ways in which control difficulties can arise via transitions in the multiloop structures are discussed in reference 3.

The time-advanced lateral deviation structure of figure 5a assumes that the driver operates on an estimated or projected lateral deviation error. Preview here is explicitly required, because this error gives the lateral position the vehicle would have at some point ahead of the vehicle if it continued along its current path (fig. 6). The range to the point of regard is given by R, which is approximately equal to velocity U_o times time T to travel to that point. Time advance T provides a perceptual preview that results in a pure lead equalization term in the effective controlled element dynamics. This, in turn, offsets the undesirable K_c/s^2 form of the lateral deviation dynamics at low frequency. When operating as described here, this system is essentially single-loop, although the multiloop aspect of figure 5a is useful for later studies of transitions and to make figure 5a comparable with figure 5b.

The amount of lead is given by R/U_o, which is under the direct control of the driver and depends simply on how far down the road he is looking. The analyses in the appendix indicate that a time of 5 to 10 seconds is adequate. Larger values are of little help under normal circumstances. Values of T less than 5 seconds (440 feet at 60 mph) are not as good because they do not compensate sufficiently for the inherent lags in the driver/vehicle system. This system structure provides a relatively high crossover frequency and good lateral position control on straight roads. Some conceptual and analytical difficulties arise when it is necessary to follow a curving roadway or a passing trajectory, e.g., a guidance law or scheme is needed to provide an appropriately advanced command input to compare with the projected lateral deviation.

The path angle plus lateral deviation system of figure 5b assumes that the driver operates on these motion quantities as separate entities. It differs from the figure 5a model by having the present lateral deviation available to the driver for comparison with a desired command input and derivation of a position error. With equivalent gains the two systems are indistinguishable in the absence of a command input. Table 5 shows that the path angle plus lateral deviation system gives adequate stability and reasonably good command following. The stability of the system is relatively insensitive to joint variations in the loop gains, although the crossover frequency and (hence) system bandwidth will change of course. Consequently, the system is reasonably forgiving of momentary lapses and does not require continuous control.

The heading angle plus lateral deviation structure of figure 5c assumes that the driver operates on these motion quantities separately and combines them to produce a steer angle response. It differs only slightly from the preceding model in its use of heading angle instead of path angle (which is heading plus sideslip) as the inner loop. Table 5 shows that the heading angle plus lateral deviation system gives good command following and performance, is

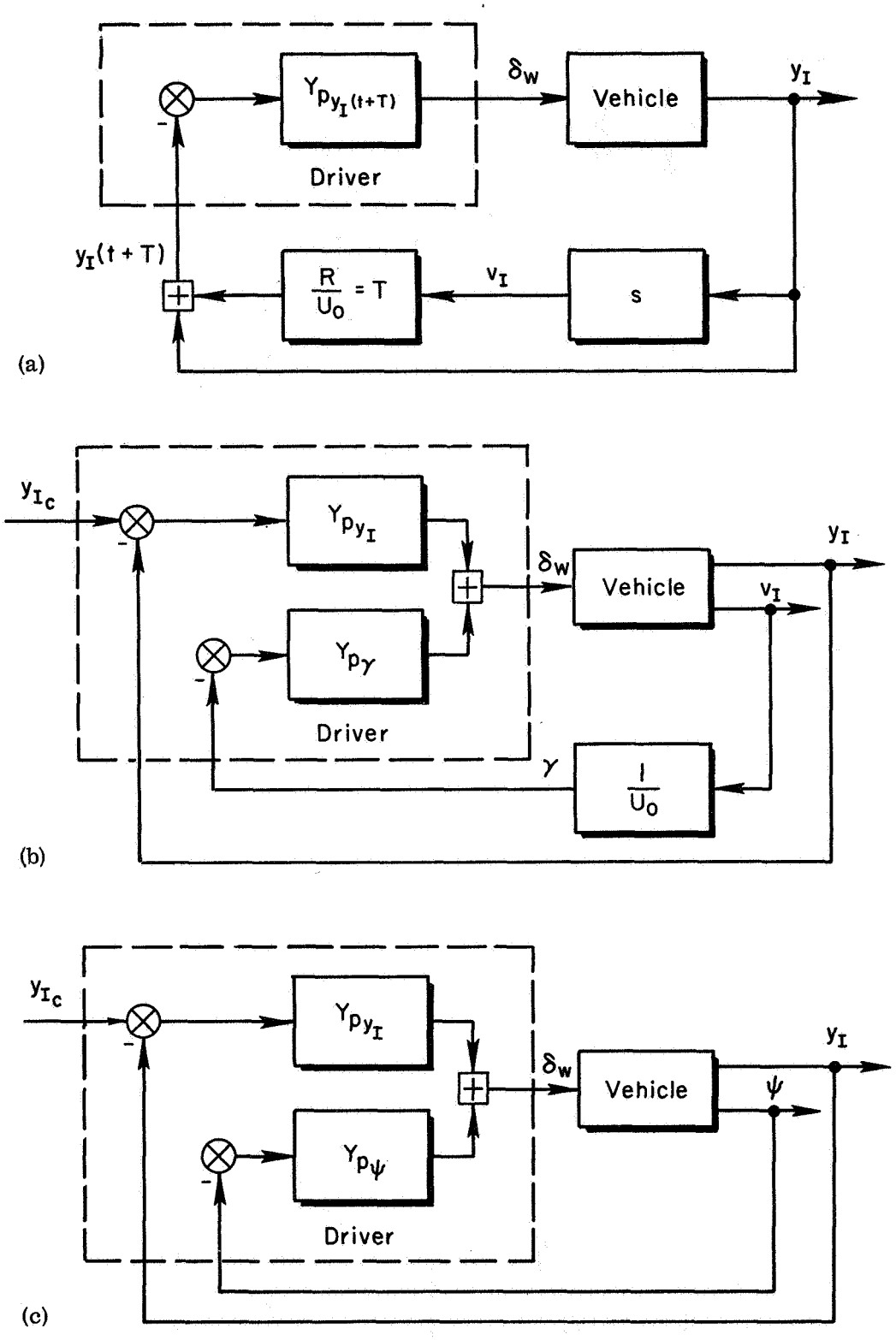

Figure 5.—Block diagrams for selected multiloop closures. (a) Inertial lateral deviation advanced in time. (b) Path angle plus inertial lateral deviation. (c) Heading angle plus inertial lateral deviation.

TABLE 5.—MULTILOOP SYSTEM CHARACTERISTICS

SYSTEM	INNER LOOP		OUTER LOOP		REMARKS
	Equal-ization	Cross-over Freq. (rad/sec)	Equal-ization	Cross-over Freq. (rad/sec)	
Inertial lateral deviation advanced in time	—	—	Gain only	1.2	Fine for straight roads. Difficult to define command input form and point of entry into the system. Sensitive to changes in preview.
Path angle plus inertial lateral deviation	Gain only	1.0	Gain only	0.53	Control not sensitive to changes in driver adaptation or driver attention. Outer loop can operate intermittently.
Heading angle plus inertial lateral deviation	Gain only	1.5	Gain only	0.6	Control not sensitive to changes in driver adaptation or driver attention. Outer loop can operate intermittently.

relatively insensitive to changes in driver adaptation, permits fairly inattentive control, and has the attribute needed for good driver opinion. It has the additional possible advantage that heading angle (which is simply the subtended angle between reference lines on the car and in the roadway) may be easier to perceive under certain circumstances than path angle (which is the angle between a roadway reference line and the line to the point of no relative motion in the surround). Note, finally, that sideslip angles are usually relatively small, in which case heading angle and path angle are almost the same quantity.

To avoid potential confusion about the relative merits of the three systems insofar as outer-loop crossover frequency is concerned, it should be emphasized that the larger value for the advanced-in-time system is an artifact of the loop closure criteria used. As already noted, the closed-loop dynamics of the three systems can be made very similar if the loop gain and preview times are appropriately adjusted. However, in making the estimates for the systems of figures 5b and 5c, the inner loops are closed with relatively large stability margins, so as to be representative of conditions with possibly intermittent closures of the outer loop. The advanced-in-time system of figure 5a, on the other hand, has no separable inner loop, so this consideration did not apply.

None of the systems shown in figure 5 include the two feedforward channels or the discrete response feedback loop of figure 1. This is because the closed-loop systems shown in figure 5 are most appropriate for command following or regulation tasks of a reasonably continuous nature with inputs that are more or less random appearing. The outer types of response involving learned maneuvers, etc., are important and do dominate the driver's control activity in some phases of driving. However, the closed-loop type of control is fundamental to many other phases of driving and plays a key take-over role in circumstances where the learned maneuvers and patterned responses either cannot be structured or are suddenly destroyed for one reason or another. The compensatory loops are also used in early phases of learning (the unskilled driver) and under conditions of extreme stress (the startled or confused driver). If the closed-loop systems will not or cannot work, then there is little chance that the driver/vehicle system will function safely for long. Thus, understanding their form and operation gives the point of departure for either adding the other channels or switching to them as needed.

These three systems do not exhaust the possible multiloop structures that can be concocted. Yet they are the only ones found to date that satisfy the guidance and control requirements for command following and disturbance regulation with good performance, insensitivity to variations in the driver's dynamic adaptation, good predicted subjective opinion from the driver, etc. Further, they are not inconsistent with available perceptual data obtained from driving experiments on the highway. In addition to modeling the driver/vehicle system in a useful way, they can also provide a new framework for devising further experiments to study driver perception and control processes. All three structures give roughly the same performance under good conditions, although they differ in some details. The principal distinction between them lies in the way in which they degrade when the nominally good structure is disturbed. Such a disturbance might disrupt the perceptual interaction between the driver and the surroundings. It might also modify the driver element by distraction, inattention, or other degradation, the vehicle element in a mechanical way, or the roadway environment's texture, illumination, etc. The way in which these transient modifications in the dynamics of elements within the guidance and control loop(s) or in the loop structure itself can lead to dangerous vehicle motions is examined in reference 3.

CORRELATION WITH EXPERIMENTAL OBSERVATIONS.—The identification of preferred driver/vehicle loop closures and sensory cues from a control theory approach

provides a new framework for reviewing past perceptual experiments. The perceptual basis is somewhat fragmentary, but all the available evidence tends to corroborate the results of analytical deduction previously cited. The following verbal evidence and remarks from representative sources will at least illustrate the trend of activities in this area.

A pioneering series of experiments by Gordon (ref. 25) attempted to define the driver's visual input experimentally. The apparatus was a helmet-mounted aperture for one eye comprised of a tube 3.5 inches long with a variable diameter of 1 inch or less. The resultant aperture angles were 4° and 9.75°. The other eye was masked. A camera was mounted on the helmet coaxially with the aperture tube. The task was to drive along a narrow, winding two-lane country road. He found that most records showed continuous visual shifts forward to the limit of the visible road and then backward toward the vehicle. This bimodality of fixation positions suggests that the driver may be looking down the road to obtain heading or path angle information, and then occasionally looking near the vehicle to sample lateral position or deviation. His data also showed that about 60 percent of the time the driver was looking farther than 150 feet ahead of the vehicle, while about 80 percent of the time he was looking more than 100 feet ahead. At the test speeds of about 15 mph (20 fps) these are preview times of 7.5 and 5 seconds, respectively. These times correlate well with desirable preview time T found for the time-advanced lateral deviation model of figure 5a. The use of small aperture viewing was reported to cause stress, suggesting that denial of the peripheral cues may lead to a significant degradation in the perceptive structure.

In a more recent paper, Gordon (ref. 26) expands on the streamer theory (after Calvert, ref. 27), which states in essence that the driver perceives motion from objects in the visual field streaming across his field of view and emanating from a central focus (when following a nominally straight path). An alternative is taken by Gibson (ref. 28) who asserts that the focus of expansion provides the directional cue rather than the flow characteristics of the velocity field emanating from the focus. Regardless of how the driver senses the direction to this focus, this cue is precisely the path angle cue discussed previously and used in the loop closures of the appendix. Gordon goes on to say that all parts of the visual field (road borders and lane markers) move when the wheel is turned, and no one part is essential for tracking. His studies showed that the driver may assume a somewhat unlocalized surveillance of the road, which would facilitate seeing a steady state (flow field) and also reduce nystagmus. He later suggests that "the driver may become aware of the car [motion] by slewing shifts in direction, and by sideslipping sidewise movements," and "sidewise movement of the road borders is perceptible either as a movement or change in position." He concludes that "on the basis of human perception theory [alone] it is difficult to determine which of the four combinations of slew, sideslip, rate, and amplitude the driver perceives. The driver responds to a total situation, not to isolated or ranked cues."

Schmidt (ref. 29) discusses perception in general, and then notes that "the driver recognizes the movement of the car from the apparent flow or streaming of the objects in the visual field. . .," agreeing in essence with Gordon and Calvert.

Biggs (ref. 30) conjectures that in driving the central vision is occupied with the detection of obstacles in the immediate path, while the peripheral vision is employed in the task of tracking the guideline or dividing line. He notes that "the guideline is seen near the vehicle, and its lateral motion provides the dominant directional cue." It could be inferred from this that the directional cue detected in the periphery from motions of the guidelines (or streamers) yields heading or path angle and possibly angular rate. Inertial guidance is, of course, just the current position of the guideline or streamer.

Even though these references are all fairly recent, none of them specifies what loops to close. They do support, however, the theory of driver control derived herein from guidance and control considerations. While there are apparent differences between these

experimental results, some semblance of order is derived by viewing them within a guidance and control structure, and the operational entities extracted from the perceptual cues do provide the prescribed feedbacks.

SOME IMPLICATIONS OF THE APPROACH

The objectives of the research of which this paper forms a part have been to gain further understanding of driver control processes, and to determine why guidance and control errors occur and how they can be avoided to reduce the hazard. The major results of the first objective have been presented here. These have involved the derivation and development of operational models for the driver/vehicle/roadway system which use the techniques and principles of feedback control theory. The resultant models provide powerful analytical tools that can be used to:

(1) Determine the role and relative importance of the various elements (driver, vehicle, etc.) in the operation of the overall system

(2) Define the levels and nature of the interaction between system elements

(3) Assess the effect on the characteristics of the overall system of changing parameters within and between the elements

In essence, a new capability has been established which can guide the highway systems engineer in his efforts to improve the driver/vehicle/roadway system, or to assess the effect on vehicle control of proposed system changes that may be thrust upon him. Although not a panacea, the approach provides fresh insights, some alternative views to the traditional, a basis for interpretation of experimental results, and a useful foundation for designing new key driver control experiments that can more fully validate the derived theory.

The second and third objectives are treated in reference 3 using example applications of the driver/vehicle closed-loop models. There the following problem areas are considered:

(1) Transitions in the driver/vehicle/roadway system, with particular emphasis on sudden changes in the perceptual structure and their varying degrees of resultant hazard. One example treated loss of driver preview in the time-advanced lateral deviation structure in the presence of a random-appearing disturbance input. Another involved failures of the path angle inner loop (while retaining the lateral deviation loop) at various points during a passing maneuver. Another example considered loss of the heading angle inner loop followed by introduction of suitable driver lead equalization in the lateral deviation outer loop.

(2) The effect of acceleration feedbacks, with emphasis on ways they can interfere with good visual loop structures. The influence of driver location (relative to the center of the vehicle) on the acceleration cue was examined first. Then transitions from visual control to steering control based on accelerations at the usual position of the driver's head were studied. Finally, inadvertent steering inputs due to arm/hand inertia of the steering wheel (in the presence of vehicle accelerations) were examined.

(3) Highway gust disturbance inputs, with attention to the types of highway gusts that can be most troublesome and how they can arise. The open-loop (steering wheel fixed) and closed-loop gust response dynamics were computed. Step and pulse crosswind inputs were modeled, and these were used to compute transient time responses of the driver/vehicle system both open- and closed-loop.

(4) Steering system dynamic lags and their significance to the closed-loop characteristics of the combined driver/vehicle system. Preliminary data were cited to show that steering lags may be as large as the driver's effective time delay. Because these delays are additive, the bandwidth and performance of the driver/vehicle system will be reduced accordingly.

The approach to these problems involved examining good driver/vehicle control structures to see ways in which they might be degraded and looking at poor situations to see how they might be improved or avoided. Many of the control concepts used (such as the adverse effect of opening an inner loop) are not new to a controls engineer, but it is their application to understanding the guidance and control of automobiles that represents an innovation. The examples shown, plus general application of the derived model to problems of this sort, provide the bases for remedial action that can help to avoid control difficulties (and reduce the hazard) or to enhance the guidance and control situation and increase traffic throughout.

APPENDIX

DRIVER/VEHICLE MULTILOOP CLOSURES.—Multiloop closures of the driver/vehicle system are appropriate to providing good command following and good regulatory performance with minimum driver effort and good subjective opinion. Three different good multiloop structures have been evolved:

(1) Inertial lateral deviation advanced in time
(2) Path angle plus inertial lateral deviation
(3) Heading angle plus inertial lateral deviation

Technical details of the closed-loop analyses needed to support table 5 of the main text are presented below, and the account is necessarily brief.

Inertial Lateral Deviation Advanced in Time.—This structure assumes that the lateral deviation along the path or track down the road can be perceived as shown in figure 6. The range to the point of regard is given by R, which is approximately equal to velocity U_o times the time to travel to point T for small angles. Similarly, R sin γ is approximated by $R\gamma$.

The corresponding block diagram is given in figure 5a. The basic vehicle dynamics are given by the $y_I \rightarrow \delta_w$ transfer function. The time advance is obtained by differentiating y_I to obtain v_I and then multiplying this by time T to the point of regard. The driver operates on the projected deviation to produce front wheel steer angle δ_w. The effective controlled element dynamics are given by

$$\mathcal{L}\left[\frac{y_I(t+T)}{\delta_w}\right] = (Ts+1)\frac{y_I}{\delta_w}(s) \tag{17}$$

where $\mathcal{L}$ denotes the Laplace transform. The effect of the perceptual preview is to provide pure lead term (Ts + 1) in the controlled element that offsets the K_c/s^2 form of y_I/δ_w at low frequency. The amount of lead is given by R/U_o, which is under the direct control of the driver and depends simply on how far down the road he is looking.

The basic $y_I \rightarrow \delta_w$ transfer function at 60 mph (88 fps) is obtained by double intregration of the lateral acceleration in table 1, i.e.:

$$\frac{y_I}{\delta_w} = \frac{180\left[s^2 + 2(0.160)(7.24)s + (7.24)^2\right]\left[s^2 + 2(0.293)(6.91)s + (6.91)^2\right]}{0.684s^2\left[s^2 + 2(0.785)(5.84)s + (5.84)^2\right]\left[s^2 + 2(0.290)(6.97)s + (6.97)^2\right]} \tag{18}$$

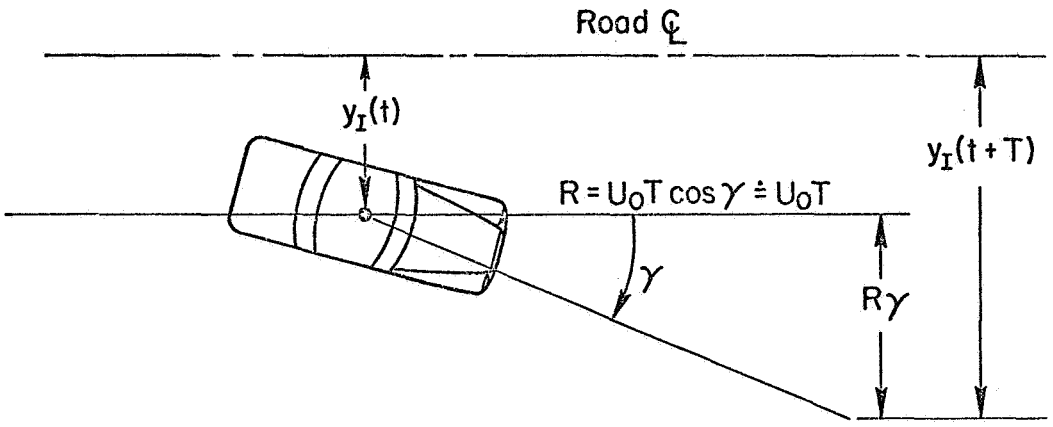

Figure 6.—Geometry for time-advanced lateral deviation.

A family of jω Bode plots is given in figure 7 for the following values of perceptual lead:

T = 1 sec = 88 ft at 60 mph

T = 2 sec = 176 ft at 60 mph

T = 5 sec = 440 ft at 60 mph

T = 10 sec = 880 ft at 60 mph

A driver time delay of 0.3 second has been included in the phase plots.

The perceptual lead creates the desirable long stretch of nearly -20 dB/decade amplitude ratio slope in figure 7. For T= 10 seconds a crossover at 1.2 rad/sec can be made with a 50° phase margin criterion (as opposed to the 60° criterion used in a later discussion). Values of T lower than about 5 seconds are not as good because they do not adequately compensate for the driver's time delay, and a good stable closure is harder to achieve. Values of T greater than about 10 seconds provide little added benefit under normal circumstances.

This multiloop structure is fine for regulation of lateral position on straight roads. Analytical difficulties arise when the model is asked to follow a command input such as curving roadway or a desired passing trajectory. As presented in figure 5a, it is necessary to construct an appropriately advanced command input via a guidance law and then to delay the driver's response by approximately the amount of the time advance. This delays the steering response of the driver/vehicle system until the previewed change in the path is reached.

Path Angle Plus Inertial Lateral Deviation.—This structure assumes that the inertial lateral deviation of the vehicle at its present position and the vehicle path angle are used as separate motion quantities. It differs from the time-advanced lateral deviation model discussed above by having the present lateral deviation available to the driver for comparison with the desired command input and derivation of a motion error. Time advance and delay guidance computations are not required. The two are indistinguishable if a command input is not present and if loop gain and preview are made compatible.

The block diagram for path angle plus lateral deviation control is given in figure 5b. A parallel structure is shown with the output of two driver blocks summing to give the net steer angle. A series structure could be employed as an alternative, with the output of the

driver's y_I control block providing an effective command at the input side of his γ control block. The differences are minor when the driver equalizations are simple gains plus a common time delay. The vehicle dynamics are given in the usual way by the transfer functions of reference 1.

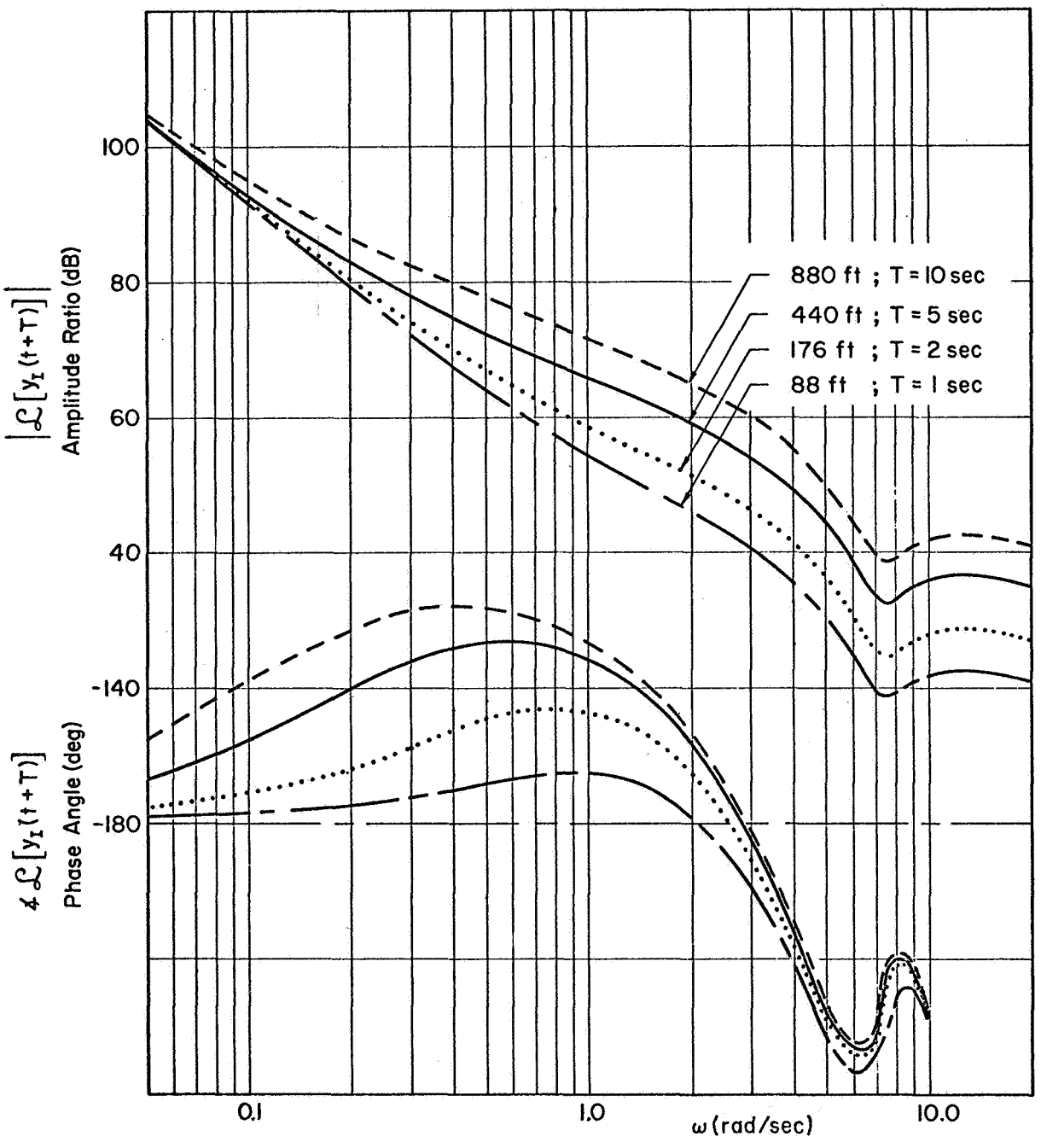

Figure 7.—Effective $y_I(t + T) \rightarrow \delta_W$ transfer function.

Assuming that the driver/vehicle task is to follow a desired passing trajectory or roadway curvature (denoted by y_{I_c}), the analytical problem is to determine the characteristics of the closed-loop driver/vehicle system. The formal procedure is to close the path angle loop and then to close the lateral deviation loop to obtain the $y_I \rightarrow y_{I_c}$ transfer function. The closed-loop function is:

$$\left[\frac{y_I}{y_{I_c}}\right]_{\substack{\gamma \rightarrow \delta_w \\ y_I \rightarrow \delta_w}} = \frac{Y_{p_{y_I}} N_{\delta_w}^{y_I}}{\Delta + Y_{p_\gamma} N_{\delta_w}^{\gamma} + Y_{p_{y_I}} N_{\delta_w}^{y_I}} \tag{19}$$

where $N_{\delta_w}^{y_I}$ is the lateral deviation numerator

$N_{\delta_w}^{\gamma}$ is the path angle numerator

Δ is the characteristic denominator

the vehicle numerators and denominator are known [i.e., ref. 1], and the driver describing functions $Y_{p_{y_I}}$ and Y_{p_γ} are to be determined. Primes are used to denote an inner-loop closure and simplify the notation, i.e.,

$$\left[\frac{y_I}{y_{I_c}}\right]_{\substack{\gamma \rightarrow \delta_w \\ y_I \rightarrow \delta_w}} = \left[\frac{y_I}{y_{I_c}}\right]''$$

$$= \frac{Y_{p_{y_I}} N_{\delta_w}^{y_I}}{\Delta' + Y_{p_{y_I}} N_{\delta_w}^{y_I}}$$

$$= \frac{Y_{p_{y_I}} N_{\delta_w}^{y_I}}{\Delta''} \tag{20}$$

The path angle (alone) system survey is given in figure 8, and this plot provides the inner-loop closure. No driver lead equalization and a time delay of 0.3 second will be assumed to provide a conservative estimate. A first-order Padé approximation to the reaction time delay is used; i.e., $e^{-\tau s} \doteq -(s-2/\tau)/(s+2/\tau)$. The driver can cross over at about 1 rad/sec on the $j\omega$ Bode of figure 8 with good stability margin, giving a driver characteristic in the path angle loop of

$$Y_{p_\gamma} \doteq 0.229e^{-0.3s} \doteq \frac{-0.229(s - 6.67)}{(s + 6.67)} \tag{21}$$

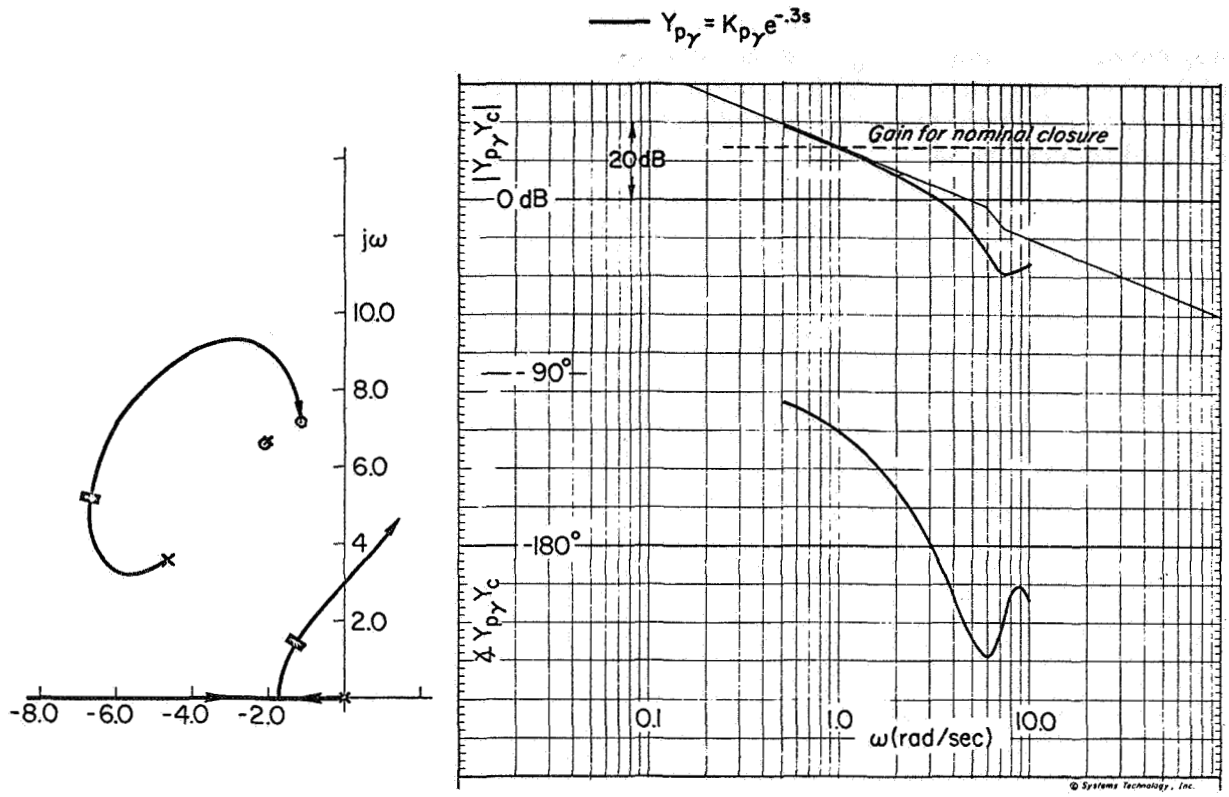

Figure 8.—Path angle loop closure.

This provides a good stable single-loop closure, and a higher gain might have been used in its role as an inner loop which would result in a higher outer-loop bandwidth. The units of equation (21) are radians of front wheel steer angle per radian of path angle error. In the notation of equation (20), the closed inner-loop denominator becomes:

$$\Delta' = \frac{.684\left[s^2+2(0.639)(1.93)s+(1.93)^2\right]\left[s^2+2(0.290)(6.97)s+(6.97)^2\right]\left[s^2+2(0.798)(7.95)s+(7.95)^2\right]}{s(s+6.67)} \tag{22}$$

The zero at -6.67 in the denominator of equation (22) results from the Padé approximation to the time delay.

The open outer-loop transfer function at 60 mph is given by:

$$\left[\frac{\gamma_I}{\delta_w}\right]' = \frac{180\not{s}(s+6.67)\left[s^2+2(0.293)(6.91)s+(6.91)^2\right]\left[s^2+2(0.16)(7.24)s+(7.24)^2\right]}{.684\not{s}^2\left[s^2+2(0.639)(1.93)s+(1.93)^2\right]\left[s^2+2(0.290)(6.97)s+(6.97)^2\right]\left[s^2+2(0.789)(7.95)s+(7.95)^2\right]} \tag{23}$$

The Bode plot for equation (23) including the driver's outer-loop time delay (assumed 0.3 second) is given in figure 9. The crossover for 60° phase margin occurs at 0.53 rad/sec and that for neutral stability at about 1.5 rad/sec. A higher bandwidth in the outer loop could have been achieved by using a higher inner-loop gain.

This combination of feedback loops provides a system with adequate stability and command-following performance. The lack of driver lead equalization means that control

will not be sensitive to the driver's adaptation nor require his close attention, and the subjective opinion should be relatively good.

HEADING ANGLE PLUS INERTIAL LATERAL DEVIATION.—The driver is assumed to perceive vehicle heading angle and inertial lateral deviation at its present position as separate motion quantities. This structure differs from the preceding multiloop closure in that it uses only heading angle for the inner loop, not heading plus sideslip which gave path angle.

The block diagram for heading plus lateral deviation control is given in figure 5c. A parallel structure for the driver is used, although a series form could be employed with only the minor differences noted previously. The vehicle dynamics are given by the transfer functions of the vehicle at 60 mph.

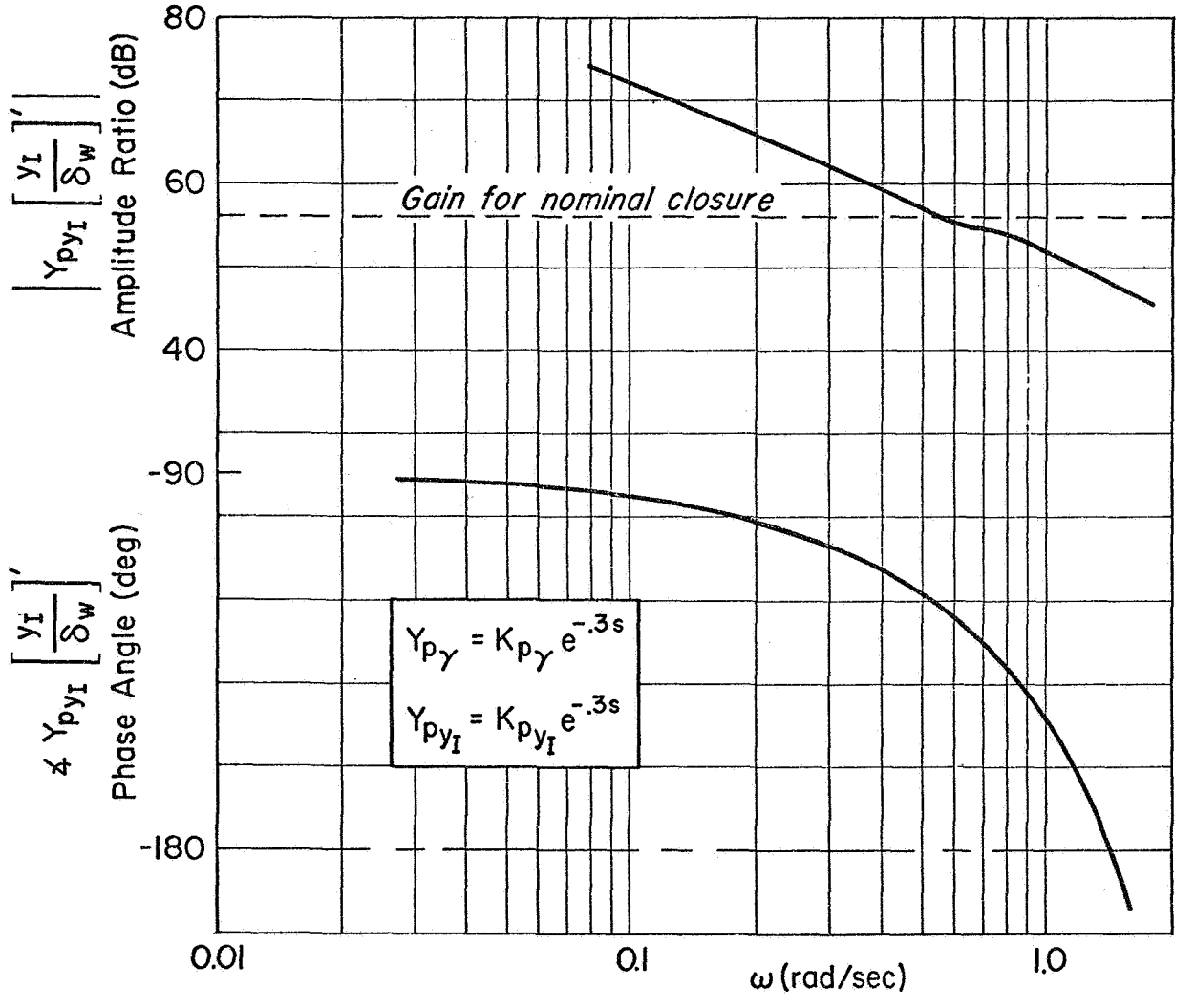

Figure 9.—Path angle plus lateral deviation closure.

The task is assumed to be the usual one of following a desired path or passing trajectory, defined as a lateral deviation command input. The closed-loop characteristics of the driver/vehicle system are to be determined by closing the heading and lateral deviation loops, successively.

The closed-loop transfer function is:

$$\left[\frac{y_I}{y_{I_c}}\right]'' = \frac{Y_{p_{y_I}} N_{\delta_w}^{y_I}}{\Delta + Y_{p_\psi} N_{\delta_w}^{\psi} + Y_{p_{y_I}} N_{\delta_w}^{\psi}}$$

$$= \frac{Y_{p_{y_I}} N_{\delta_w}^{y_I}}{\Delta''} \tag{24}$$

where $N_{\delta_w}^{\psi}$ is the heading angle numerator and the notation is that of equation (20).

The system survey for heading angle alone is given in figure 10, and this provides the basis for closing the inner loop in this multiloop case. The driver is assumed to have a pure gain plus time delay (0.3 second) form. A conservative crossover with good stability margins is 1.5 rad/sec, giving a driver heading response characteristic of:

$$Y_{p_\psi} \doteq 0.32e^{-0.3s} \doteq \frac{-0.32(s-6.67)}{(s+6.67)} \tag{25}$$

The units of equation (25) are radians of front wheel steer angle per radian of heading error. The closed inner-loop denominator becomes:

$$\Delta' = \frac{.684(s+2.664)(s+9.397)[s^2+2(0.524)(3.61)s+(3.61)^2][s^2+2(0.289)(6.99)s+(6.99)^2]}{s(s+6.67)} \tag{26}$$

The zero at -6.67 in the denominator of equation (26) results from the Padé approximation, as before.

The open outer-loop transfer function for lateral deviation at 60 mph is:

$$\frac{y_I}{\delta_w} = \frac{180\,s(s-6.67)[s^2+2(0.293)(6.91)s+(6.91)^2][s^2+2(.16)(7.24)s+(7.24)^2]}{0.684\,s^2(s+2.664)(s+9.397)[s^2+2(0.524)(3.61)s+(3.61)^2][s^2+2(0.289)(6.99)s+(6.99)^2]} \tag{27}$$

The Bode plot for the effective outer-loop vehicle of equation (27) plus the driver's time delay (0.3 second) in the outer loop is given in figure 11. In addition to the nominal heading loop gain ($K_{p_\psi} = 0.32$) case, the $(y_I \rightarrow \delta_w)'$ transfer functions are shown for several other heading loop gains. The nominal gain gives a lateral deviation loop crossover at 0.6 rad/sec with 60° phase margin. A slightly higher gain ($K_{p_\psi} = 0.372$) gives a slightly better outer-loop system (crossover at 0.68 rad/sec) with adequate gain margin and the same phase margin. The differences are minor, however, and the nominal case was chosen to provide comparisons among the various loop structures with the same closure criteria.

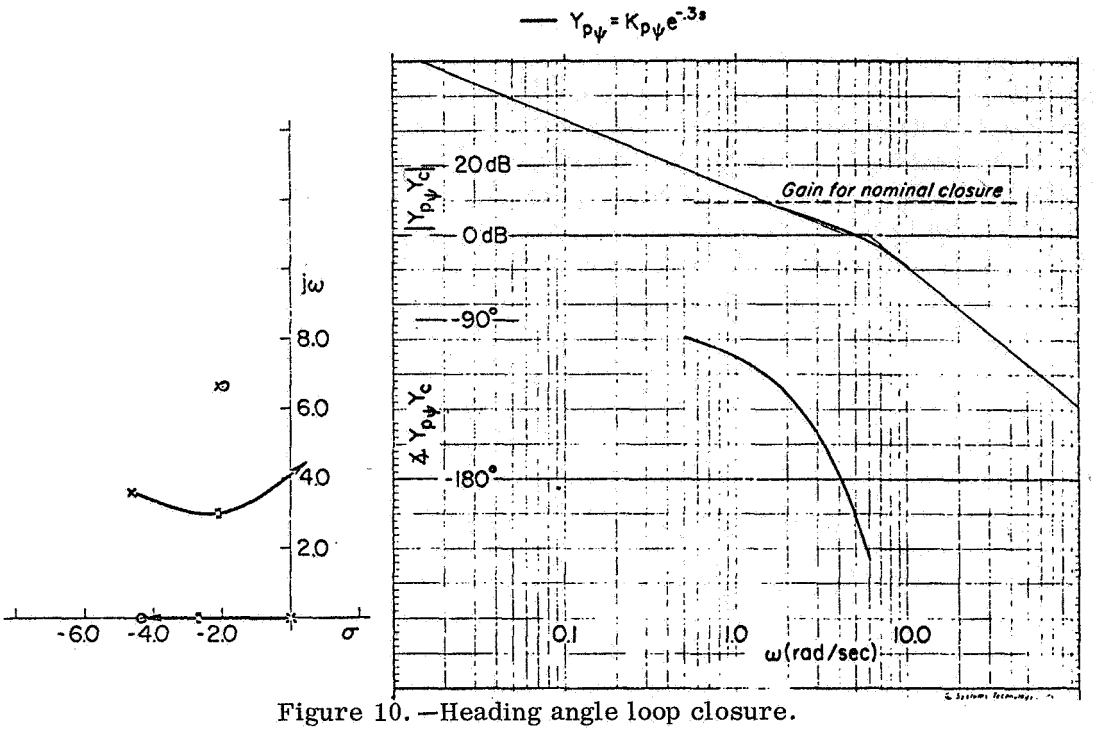

Figure 10.—Heading angle loop closure.

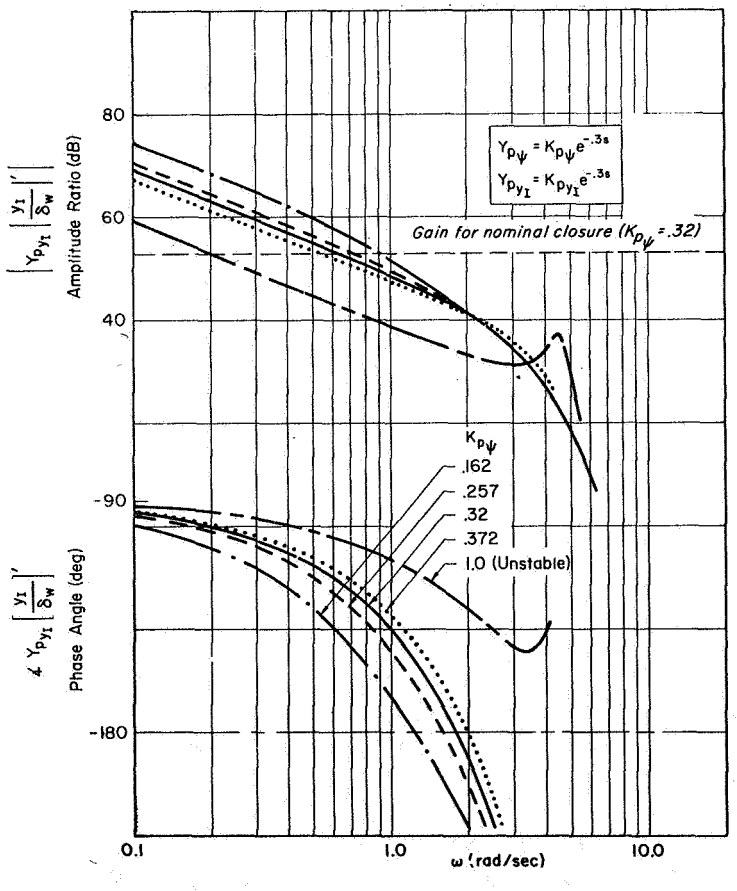

Figure 11.—Heading angle plus lateral deviation closure.

A large increase in heading loop gain results in a poor (and ultimately unacceptable) outer-loop system. This is illustrated in figure 11 for $K_{p_\psi} = 1.0$, approximately three times the nominal value. This value of gain makes the low frequency quadratic pair in Δ' unstable (it has a negative damping ratio), and closing the outer loop simply makes this pair more unstable. The peaking of the outer-loop amplitude ratio due to a high inner-loop gain is illustrated by the $K_{p_\psi} = 1.0$ case. Even if the damping were positive (but small, say, $K_{p_\psi} \doteq 0.9$) the crossover would have to be at a relatively low frequency to leave at least a 6-decibel gain margin between the top of the peak and the loop closing the 0-decibel line.

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